Coupled rolling and buckling model for friction-sensitive setting of flatness actuators

Rebecca Nakhoul [a], Sami Abdelkhalek [b], Pierre Montmitonnet [a]

[a] MINES ParisTech, CEMEF, UMR CNRS 7635, Sophia-Antipolis, France
[b] ArcelorMittal R&D, Maizières-les-Metz, France

Abstract

Flatness defects in thin strip cold rolling are a consequence of buckling due to roll thermo-elastic deformation and the resulting heterogeneous strip plastic deformation. A coupled rolling – buckling model has been developed [Abdelkhalek et al. 2011]. It has shown that (i) taking buckling into account results in completely different stress fields and fits correctly the measured on-line residual stresses under tension ("stress-meter rolls"); (ii) coupling buckling in the post-bite area and the rolling model, whatever the technique, changes little the in-bite fields. The model is applied here to the effect of friction on optimal setting of a flatness actuator, Work Roll Bending.

Coupled rolling - buckling models

Sheet Rolling model Lam3/Tec3

The rolling model is Lam3/Tec3, a 3D strip / roll stack deformation software described in [Hacquin et al.1998]. It uses an implicit velocity formulation with P1-discretisation on hexahedra. A *steady state* formulation based on streamlines is used, which requires EVP constitutive equations integration along streamlines, with an heterogeneous time step strategy called ELDTH.

The roll stack deformation model is based on advanced beam theory, Boussinesq solution of a half-space under general loading, and Hertz contact mechanics. This model is discretized by an influence function method, resulting in a system of equations in the roll rigid body displacement, contact line displacement field roll / roll and contact pressures. This system is non-linear due to unknown contact lines. It is solved by Newton-Raphson method.

Counhaye-Abdelkhalek model

[Counhaye, 2000] has proposed a method to deal with sheet buckling in a FDM rolling model. The same has been implemented in Lam3/Tec3 [Abdelkhalek 2011]. Initially proposed within the membrane theory framework, it forbids the appearance of a negative stress: every time this is about to occur, the structure buckles, bringing the stress back to almost zero by providing a stress-free alternative to *elastic* shortening of a material line. The following critical conditions are introduced:

$$\begin{aligned} &\vec{n}_1.\sigma.\vec{n}_1 = 0 \\ &\vec{n}_2.\sigma.\vec{n}_2 > 0 \\ &\vec{n}_1.\sigma.\vec{n}_2 = 0 \end{aligned} \qquad (1)$$

$\vec{n}_1$ and $\vec{n}_2$ are the directions of the principal Cauchy stress tensor in the buckled structure. This means that when a tension is applied in a direction (here $\vec{n}_2$), the membrane is stiff; if the stress becomes negative, it gets slack and the corresponding stress is put to 0 (direction $\vec{n}_1$). The essence of the model consists in determining an extra deformation which elastically brings the stress in the buckled direction back to 0. It may be interpreted as the shortening of a material line due to buckling of the structure. This is more or less analogous to elastic-plastic decomposition, but is activated only out of the roll bite, i.e. where buckling may manifest

$$\Delta\varepsilon = \Delta\varepsilon^{el} + \Delta\varepsilon^{bu} \qquad (2)$$

where $\Delta\varepsilon^{el}$ is the elastic and $\Delta\varepsilon^{bu}$ is the "buckling strain" increment. Plane stress is assumed out of bite.

The extra deformation representing buckling is computed in the principal axes then transferred to the reference frame. Let λ_i, i = I, II, be the deformation representing buckling in the principal directions. It is deduced from σ_i, i = I, II as follows:

$$\lambda_i = \frac{\langle \sigma_i - \sigma_c \rangle}{E} \quad i = I, II \qquad (4)$$

Moving back to the reference frame, the buckling strain is added to the global strain increment (u and v are the two in-plane incremental displacements, θ is the angle between principal and reference frames, ν is Poisson's ratio and E is Young's modulus), and equation (2) becomes:

$$\begin{aligned} \Delta\varepsilon_{11} &= \frac{\partial u}{\partial x} + \lambda_I \cos^2\theta + \lambda_{II}\sin^2\theta \\ \Delta\varepsilon_{22} &= \frac{\partial v}{\partial y} + \lambda_{II}\cos^2\theta + \lambda_I \sin^2\theta \\ \Delta\varepsilon_{12} &= \frac{1}{2}\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) + \left|\lambda_{II} - \lambda_I\right|\cos\theta\sin\theta \\ \Delta\varepsilon_{33} &= -\frac{\nu}{1-\nu}(\varepsilon_{11} + \varepsilon_{22}) \end{aligned} \qquad (5)$$

This strain increment replaces the standard one fed into the module which solves the constitutive differential equations.

Friction-sensitive WRB setting

The example shown hereafter refers to the last stand of a tinplate sheet mill, with very low thickness (0.252 mm at exit); the width is 855 mm. The stand is a 4-high one.

Work Roll Bending (WRB) is a typical sheet profile and flatness actuator: by applying a torque opposed to the contact stress moment, rolls are brought closer to their rest shape. This is a more flexible actuator than e.g. roll crown, which is chosen once and for all and fixed. The WRB force can be controlled using monitoring of the stress profile by the stress-meter roll. The question of its variations along a coil is therefore of importance. Friction may vary, due to (i) progressive roll wear from coil to coil and (ii) lubrication efficiency changes during accelerations and deceleration at coil beginning and end. It is important to quantify the effects of such friction variations.

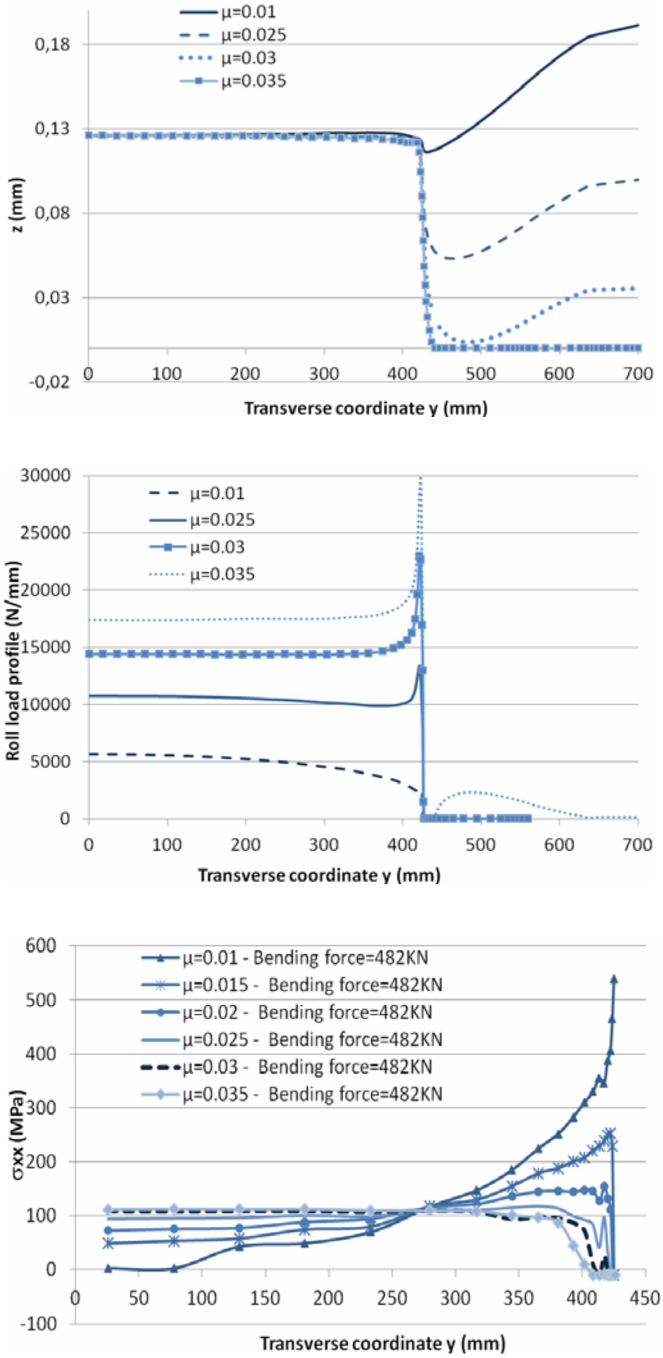

Figure 1 *The impact of friction on work roll generator shape (top), work roll load profile (middle) and transverse profile of strip stress (latent flatness defect, bottom)*

In figure 1, a parametric study is described: friction varies between μ = 0.01 (slight skidding) and μ = 0.35. The impact on roll deformation under a WRB force of -482 kN is first shown: low friction gives a low rolling load, and moderate roll deformation, the shape of the WR generator therefore mainly corresponds to counter bending by the WRB force. At the highest μ, the rolling load is very high (compare the WR load profiles in the middle) and roll kiss occurs (WRs touch each other on either side of the strip).

The strip reduction profiles are therefore quite different, and so are the residual stresses (bottom picture): μ = 0.01 gives strong tension on the edges and a slack centre (wavy centre), whereas the stress profile is most flat for μ = 0.03.
On the contrary, the WRB force is varied at constant μ = 0.025 in figure 2. The very high WRB force (900 kN) again gives a low stress in the centre, which will probably result in a wavy centre, at least after tension is cancelled. The most flat stress profile seems to be with a bending force of 350 kN.

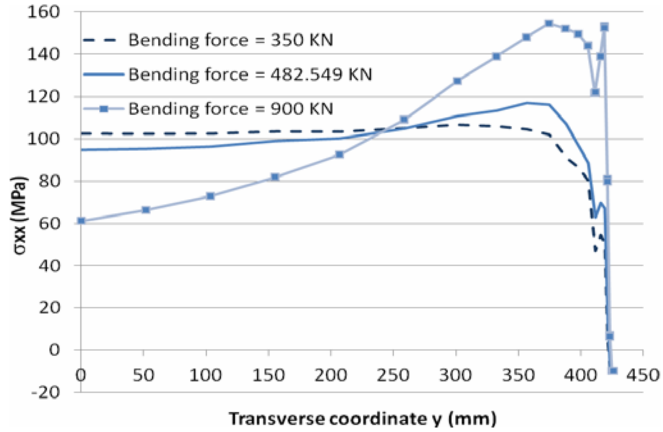

Figure 2 *The $\sigma_{xx}(y)$ stress profile for varying bending force and fixed friction coefficient (μ = 0.025)*

Finally, figure 3 summarizes the bending force found "optimal" for each value of the friction coefficient, i.e. giving the most flat stress profile "by eye". This graph gives an idea of how to pre-set the WRB force as a function of varying friction.

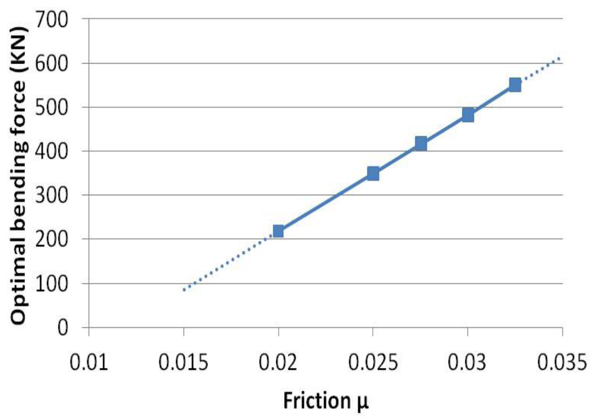

Figure 3 Relationship between friction and optimal bending force

References

A. Hacquin, P. Montmitonnet, J.P. Guilleraut, A steady state thermo-elastoviscoplastic finite element model of rolling with coupled thermo-elastic roll deformation, J. Mater. Proc. Technol. 60, 1998, 109-116

S. Abdelkhalek, P. Montmitonnet, N. Legrand, P. Buessler, Coupled approach for flatness predictions in thin strip cold rolling, Int. J. Mech. Sci. 53, 2011, 651-675

C. Counhaye Modélisation et contrôle industriel de la géométrie des aciers laminés à froid. PhD Thesis, Liège University, 2000 (in French)